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TECHNICAL NOTE

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A VISUAL STUDY OF TWO-PHASE FLOW IN A

VERTICAL TUBE WITH HEAT ADDITION

By Yih Yun Hsu and Robert W. Graham

Lewis Research Center
Cleveland, Ohio

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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SUMMARY

A visual study of two-phase flow with heat addition from the wall has been performed in two glass tube geometries. The flow regimes of bubbly, slug flow and transition from slug to annular flow were recorded by a high-speed motion-picture camera. Appreciable differences between adiabatic two-phase flow or two-component flow and nonadiabatic two-phase flow were observed. These differences emphasize the interaction of the heat transfer and the hydrodynamics in two-phase flow and the inadequacy of adiabatic studies in the accurate portrayal of the flow model.

INTRODUCTION

In recent years, two-phase convective heat transfer has been the subject of much research. Numerous investigators have conducted experiments in which various correlation schemes have been applied. Reference 1 contains an excellent review of the state of the art of boiling-convective heat-transfer correlations and their applicability to experimental information.

Despite considerable progress during the last decade, no generalized correlation of two-phase convective heat transfer exists, and there is considerable ignorance concerning the mechanism of heat transport in this peculiar fluid regime. In two-phase flow, various distributions of liquid and vapor across the channel are realized depending upon geometric, fluid flow, and heat-flux conditions. These complex variations in the distribution of the two phases make it difficult to formulate a model (or models) of heat transport.

One fruitful area of experimental research has been devoted to finding out what two-phase flow looks like in a channel. Such visual studies of the hydrodynamics would presumably be helpful in developing the models of fluid flow necessary for any analytical treatment. Flow visualization has been performed in transparent test sections where two-phase mixtures of a fluid have been introduced or where mixtures of gas and a liquid have been injected to simulate two-phase flow. In either of these situations no heat was added to the fluid in the transparent section. Thus the pictures of two-phase flow or simulated two-phase flow represent conditions close to the adiabatic case. The adiabatic studies have been valuable in classifying two-phase flow into a number of regimes. A very complete bibliography on studies of two-phase flow patterns is presented in reference 2. One of the investigations mentioned therein (ref. 3) suggested three categories in the order of increasing quality needed to produce the flow condition. They are:

(1) Bubbly flow, in which discrete vapor bubbles (emanating from the hot wall) are dispersed into the main liquid flow

(2) Slug flow, where alternating regions of vapor or liquid pass a given point in the pipe or channel

(3) Annular flow, where a thin layer of liquid is established along the wall and the core of the flow is a dispersion of liquid in a vapor matrix

Later, a fourth category of two-phase flow called mist flow was classified for the extremely high quality cases beyond annular flow. The thin liquid layer adjacent to the wall for annular flow is destroyed, and a dispersion of liquid droplets in a vapor is the flow model.

Different definitions of these flow regimes have been suggested by several investigators. However, there is sufficient agreement among all these suggestions to make the previous categories generally acceptable. In general, these categories of two-phase flow have also been observed when a simulating liquid-gas mixture has been employed. Reference 4 categorizes liquid-gas mixtures into four flow regimes according to the nature of the pressure-drop characteristics accompanying each type, and their postulated flow regimes are compared with several other catalogs of two-phase flow regimes.

In references 5 to 7 gas-liquid mixtures were used to study the hydrodynamics of two-phase flow. It was suggested that there is an analogy between "flooding" in countercurrent flow and transition between regimes in cocurrent flow. The correlating parameters (ref. 6) are similar to the velocity parameters used in the design of packed-bed towers. By use of the correlating parameters, which are dimensionless velocities, it was shown (ref. 6) that the suggestion had considerable merit. Comparison with experimental results did reveal that slug and annular flows in vertical tubes occurred at set values of the dimensionless gas velocity. In another interesting study with two-component flow (ref. 7), the air was injected through the walls of the channel rather than by an upstream injector or mixer. The flow regime was found to be very sensitive to the flux of air introduced in this fashion. This means of introducing air was intended to simulate the effect of bubble ebullition on two-phase flow.

While the adiabatic studies of the hydrodynamic models of two-phase flow do provide some useful information, it would be highly desirable to visualize the flow when heat is being added to the fluid. This is the situation of most interest to two-phase-flow applications. Certainly, such factors as bubble ebullition and acceleration of the flow caused by changes in quality that are associated with heat addition will influence the nature of the two-phase flow. A visual study of two-phase flow in an annulus (inner tube heated) is presented in reference 8.

Contained herein are the results of a visual study of two-phase flow in heated glass tubes. A thin conducting film was deposited on the exterior surface of the tubes, and the tubes were heated by electrical resistance heating. The conducting film was thin enough that the transparency was impeded only slightly. Two 30-inch-long tube geometries were studied; one had an external diameter of 13 millimeters and the other 19 millimeters. The mass flow through the tube was maintained at a relatively constant value for all tests, but local variation in fluid velocity from 0.25 to 20 feet per second reflected variations in the void fraction. The heat flux was varied from the adiabatic condition to 0.1 Btu per square inch per second. Bubbly, slug flow and the transition from slug to annular flow were produced in the tubes. The hydrodynamic models observed will be compared with those observations made with the isothermal studies of two-phase and gas-liquid mixtures.

A motion-picture film supplement has been prepared and is available on loan. A request card and a description of the film are given in the back of the report.

The authors are indebted to Mr. David G. Evans of the Lewis Research Center who developed the apparatus on which this study was carried out.

APPARATUS AND PROCEDURE

Figure 1 is a schematic diagram of the test apparatus. The system was essentially a pump-fed loop for circulating water through the glass test section. The loop had provision for preheating the water to a predetermined temperature before it was admitted to the test section. A heat exchanger employing alcohol as a coolant served as a condenser and cooler to remove enthalpy from the fluid when it left the heated test section. By regulating this heat exchanger, the flow rate, and the thermal control on the water supply tank, a steady-state, continuous flow to the test section could be maintained. The vertically mounted test section was a glass tube with a thin metallic coating on the outside surface so that the tube could be Joule heated. The test section was subdivided into four segments (fig. 1), and a parallel hookup was employed to heat the segments. This arrangement was used to better match tube resistances to the voltages of the power supply and thus to maximize the overall power to the tube. In subdividing the tube, care was exercised in locating the electrodes so that the resistances in each segment were known precisely. The high-speed movie camera was mounted on a tripod, and the camera was positioned adjacent to the segment of the test section to be observed.

The water in the supply tank was heated to a temperature approximately 30° below the saturation temperature. When the water assumed the desired temperature, a flow rate was set by controlling the throttle valve. The secondary alcohol flow loop for the heat exchanger at the discharge of the test section was started. A mixture of dry ice and alcohol served as the heat sink for this heat exchanger. Next the electrical power to the test section was turned on. After equilibrium conditions were maintained, the following measurements were made:

(1) Electrical power to the test section

(2) Flow rate

(3) Surface temperature at two locations on the test section

(4) Inlet bulk temperature and pressure

(5) Exit pressure

When desired, a high-speed motion-picture camera recorded the flow pattern. Generally, the camera was run at 2000 frames per second. The procedure was repeated over a range of heat fluxes. The upper limit of the heat flux was established by the maximum tube surface temperature (660° F) allowed by the thin conductive coating. For all the tests, the weight flow was held constant. Two test-section diameters (13 and 19 mm) were employed, which produced two levels of bulk velocity for the constant weight flow.

RESULTS AND DISCUSSION

Description of Visualization

High-speed motion pictures were taken of the two-phase flow regimes during the course of the research program. Some of these films are presented as a narrated film supplement to this report. The flow patterns observed in this work are in the regimes of bubbly flow and slug flow. A transition pattern from slug to annular flow was observed at the greatest heat flux, but a truly annular case was not achieved. In general, the flow patterns appear to be similar to those of two-phase flow without heat addition. Figure 2 shows enlargements of frames taken by the high-speed camera in the bubbly- and slug-flow regimes. However, a few important features are observed that should be considered unique to the two-phase flow with evaporation. In order to facilitate discussion, the bubbly- and slug-flow regimes will be discussed separately. In each flow regime, the discussions will be broken down further into the cases of an active heating surface with a large number of active nucleation sites and a passive heating surface, which had only a few active sites.

Bubbly Flow

Active heating surface. - In the bubbly flow, new bubbles form frequently on the heating surface. These bubbles depart from the heating surface with high velocities and have high enough momentum to penetrate deep into the mainstream. Frequently, bubbles were observed to penetrate across the whole flow field to the other side of the tube. This was observed more frequently for the case of the smaller diameter heating tube (13 mm). A typical bubble trajectory is shown in figure 3. Thus the ratio of the penetration distance y_p to the tube diameter D_t and the stream velocity play a very important role in determining the spatial distribution of bubbles. For a heated surface, the bubble distribution is determined by bubble trajectory in addition to the diffusion mechanism and the "lift" force imposed by the flow. The last two effects control bubble distribution in a bubbly flow without ebullition (ref. 9). When a bubble is growing and departing from the wall, the surrounding flow could be so disturbed that those bubbles already in the flow would have to alter their courses in very abrupt ways.

Passive heating surface. - Before the heating surface was aged, there were less active nucleation sites on the heating surface of the small-diameter tube than that of the large tube. Consequently, in the bubbly-flow regime, the bubbles formed on the wall were fewer and larger. When the bubbles started to merge, sometimes a large bubble about the size of the tube diameter would form. Apparently, this occurred at the verge of the slug-flow regime. In other words, the bubbly-flow regime was not very apparent in this case. This will be discussed in more detail in the next section.

Slug Flow

Active heating surface. - When there were large numbers of bubbles distributed in the flow because of ebullition from the wall, the bubbles began to merge and started the slug-flow regime. Because of the continuous formation of bubbles on the heating surface, the liquid slugs contained a large number of bubbles and the head and tail of the slugs were not well-defined. The pictures gave the impression that a liquid slug had an advanced party of liquid droplets onrushing into a vapor void. In fact, an all-liquid slug never made its appearance. It looked more like a bubbly mixture. The bubbly mixture gradually transitioned into a vapor void without a definite head or tail. When the void moved upward in the direction of flow, a liquid film adjacent to the wall started to flow downward. A capillary ripple formed at the liquid-vapor interface because of the velocity difference between the two phases. This kind of ripple has also been observed in the adiabatic slug flow. In this case of boiling slug flow, however, another kind of disturbance was observed on the liquid film; it consisted of ring-shaped ripples caused by the growing and collapsing of active bubbles. These bubbles appeared to be bigger than those that form inside a thick layer of liquid. When they collapsed, a ring-shaped ripple propagated radially from the center of the bubble into the liquid film. A good analogy for these two classes of waves would be the following: the windblown ripples on a stream are the capillary type and the ripples caused by the dropping pebbles into a stream are the bubble-ebullition type. Presumably, the latter additional disturbance would enhance the heat-transfer coefficient, especially in the vicinity of the active sites.

It was reported in reference 9 that no bubbles formed on the liquid film with slug flow. For the study reported herein, bubbles were observed forming in the liquid film during most of the slug-flow runs. It is postulated that bubbles will form unless one or both of the following conditions exist:

(1) There is a lack of nucleation sites on the heating surface.

(2) The liquid film is so thin and, consequently, the laminar sublayer is also so thin that most of the nucleation sites are inactive (ref. 10).

Slug flow in a passive tube. - As was mentioned previously, the smaller tube (13 mm) had few active sites. The slug flow developed without much bubbly flow preceding it. In fact, when the tube was brand new, it was observed that the heated fluid would be flowing in a single liquid phase for a long section, then momentarily a very long Taylor bubble would develop and flow upward. Then single-phase liquid flow resumed until another long bubble formed. This process could repeat for quite a while. Even after the tube had been aged, only a few large bubbles were formed, and then they immediately merged and grew into a long Taylor bubble. The growth of this kind of Taylor bubble was so fast it resembled the cavitation phenomenon. On liquid film, either no or only a few nucleations took place. One possible explanation is that, because of the scarcity of nucleation sites, a large amount of heat was stored in a very thick and thermodynamically unstable, superheated, thermal layer. When the few bubbles were formed on the nucleation sites, they merged and formed a large bubble into which the superheated thermal layer released its excess energy by vaporization. Thus the bubble grew very rapidly into a long Taylor bubble. It could be considered as a two-stage nucleation process. This class of slug flow is herein called "cavitation slug flow" to distinguish it from the regular slug flow caused by the coalescence of numerous bubbles.

This observation demonstrates the importance of surface condition in establishing the two-phase flow pattern. If a tube surface lacks a large number of effective nucleation sites (ref. 10), there will be either no or very little bubbly-flow regime, and the flow will go from essentially a single-phase liquid flow directly into a slug flow.

Acceleration and Pulsation of Flow

Strong pulsations of the flow were observed in some of the runs. Apparently, the pulsation was started mostly by the fluctuation of pressure in the pump and was amplified through the resonance in the flow. During pulsation, the bubbles in the flow would contract and expand in response to the pressure fluctuation, and sometimes ebullition processes on the wall would start and stop in response to pressure fluctuations. These periodical changes in bubble size and ebullition cycle would serve as feedback to amplify the fluctuation and cause flow instability. This problem has been encountered by numerous engineers operating a boiling two-phase loop.

Another evident feature in the two-phase flow with heat addition is the acceleration of the flow in the core. This is especially true with the motion of long Taylor bubbles, but even the liquid slugs appeared to be accelerating as they proceeded downstream. The acceleration of the flow is attributed to the increase in volume flow rate. Conceivably, this accelerating flow would affect the boundary layer.

Penetration of Bubbles in Bubbly Flow

As has been discussed previously, adiabatic and nonadiabatic slug flows can be markedly different because of the formation of bubbles on the wall. When these bubbles grow and depart, the momentum enables the bubbles to travel in a direction transverse to the mainstream velocity. It would be possible to determine the trajectory of bubbles and, consequently, predict their spatial distribution by a balance of initial momentum against dissipation of momentum due to drag in the radial and longitudinal directions. This calculation would require information about the drag characteristics of bubbles and eddy diffusion of two-phase flow. Such an analytical approach does not seem feasible at this time without additional information.

One possible approach is to consider the analogy between the wake of a sphere and a free jetstream. A similar analogy has been used previously on pool nucleate boiling in estimating heat-transfer coefficients (ref. 11). For bubbly flow, a bubble ejecting normally into the mainstream can be considered analogous to the normal discharge of jets into a mainstream. The latter problem has been studied by many researchers (ref. 12).

It has been found (ref. 12) that, for the case of a jet discharging into a stream, the important parameters are y/D_j and $(v_j/u_\infty)\sqrt{x/D_j}$ where D_j and v_j are the diameter and velocity, respectively, of the jet at the orifice, x is the distance in the direction of flow, and u_∞ is the mainstream velocity. (Symbols are defined in the appendix.) The trajectories of jets can usually be correlated by these two parameters. For the case of ejecting bubbles, the corresponding parameters should be y/D_b and $(v_b/u_\infty)\sqrt{x/D_b}$ where D_b and v_b are the diameter and velocity, respectively, of bubbles at departure.

The analogy between jet penetration and bubble penetration is shown in figure 4. From the high-speed movies, the bubble trajectories were measured together with D_b, u_∞, and v_b for several runs. The magnitude of v_b was considered to be the bubble-growth velocity measured just before the bubble departed from the wall. The bubble width at that moment was taken as D_b. The mainstream velocity was measured by following the path of some minute bubbles flowing in the mainstream. These minute bubbles are so small that it could be assumed to be essentially flowing with the mainstream without much buoyancy velocity with respect to the stream. The parameters y/D_b and $(v_b/u_\infty)\sqrt{x/D_b}$ were plotted in figure 5, where they are referred to as the position and velocity parameters, respectively. The conditions for the various bubbles in figure 5 are listed in table I. Also shown on the plot is the solid line, which represents the curve for the discharging jets (fig. 4 of ref. 12). An analogy between the ejecting bubbles and discharging jets does exist. This analogy can be applied to the steady-flow situation only and does not apply to pulsating flow. It is believed that if the bubble trajectories are predictable, the void distribution due to the addition of bubbles from the wall could be determined, provided the population of nucleation sites was known.

Mapping of Two-Phase Flow Regimes

Because of the inadequacy of data, no attempt was made to map the various flow regimes completely. The flow regimes covered by the present data are bubbly flow and slug flow. It would be interesting to compare the result with the mapping of comparable flow regimes for nonheating two-phase flow.

One of the most generally used ways of mapping the nonheating two-phase flow regimes is based upon the parameters of volumetric flow ratio $V_v/(V_l + V_v)$ and Froude number $(u_l + u_v)^2/Dg$ (ref. 13). A flow-regime map from reference 13 is reproduced in figure 6. On the map also are shown the symbols representing data from the present study. This preliminary result shows that probably a new map is needed for the case of nonadiabatic two-phase flow. The scheme used in figure 6 for mapping the regions for two-phase flow is limited in a number of ways, one of which is that it does not include the effect of the fluid properties. This is a serious omission, but it is difficult to relate the boiling mechanism of a fluid to its equilibrium properties. Even small amounts of impurities in water will change the character of the boiling. Consequently, it does not seem reasonable to assume that a universal map of flow patterns can be generated that applies to several fluids, or even to one fluid in which the thermodynamic state is changing. In the present study, the volumetric flow rate used for mapping purposes was based upon the assumption of thermodynamic equilibrium. This assumption has been used widely in estimating the steam quality and the flow rate in two-phase flow problems. Although the equilibrium assumption is incorrect, such an assumption had to be made because of a lack of information as to the true quality and slip ratio.

According to reference 6, the transition from slug flow to annulus flow would occur when $u_v^* = 0.525$. In the present study, the highest u_v^* achieved was 0.417 and represented the slug flow pattern; thus there is no discrepancy between the experimental result and the prediction in reference 6.

EFFECT OF HEAT FLUX ON FLOW PATTERN

In simulated boiling two-phase flow (ref. 7), the flow pattern is strongly affected by the rate of addition of the gas phase from the wall. To check whether a similar situation exists in two-phase flow with heat addition, experiments were carried out with constant mass-flow rate (0.035 lb/sec) and total heat input rate (1.52 Btu/(sec)) but with various lengths of heating section. The inlet temperature (180° F) and the pressure (1 atm) also remained constant. The results of these experiments at these conditions are listed in the following table:

Run	Heating length, in.	Average heat flux, q_{av}	Local heat flux at point of observation, q_{local}	Location of section photographed, in. from inlet	Observation
A	30	Low	Low	28 - 30	Well-developed slug flow
B	27	Medium	Medium	25 - 27	Slug flow with occasional bubbly flow
C	24	High	High	22 - 24	Bubbly flow with bubbles coalescing
D	24	High	Zero	25 - 27	Bubbly flow developing into slug flow

The trend in runs A, B, and C showed that, for the same flow rate and heat input, the flow pattern could vary from bubbly flow to slug flow as the heating length increases and the average heat flux decreases.

The flow pattern may be determined either by the heating length (i.e., the length required to develop into a flow pattern) or by heat flux, or both. Runs C and D were taken with the same average heat flux but at different sections along the tube, and the flow pattern in D is more advanced toward slug flow. Thus at a certain quality, even if the slug flow may be the preferred configuration, it still takes a finite length to develop into the slug pattern. This observation checks with the findings of reference 14.

On the other hand, comparison of B and D suggests that the length requirement for the full development of a flow pattern is not the only factor. In these two runs, the locations were the same and the total heat inputs were the same. However, the upstream average heat fluxes and the local heat fluxes were different. In addition to the length effect, the heat flux might have an effect on flow pattern, too. The exact effect of heat flux on the flow is difficult to assess. One obvious effect would be on the ebullition process, since the bubbles growing on the wall may stretch into the mainstream to intercept the oncoming bubbles and thus promote coalescence. But all the factors, such as the change of pressure due to ebullition, the causes of flow pulsation, and the mixing action of injecting bubbles, and so forth, would affect the flow pattern.

If the incoming flow is subcooled, the situation is more complicated. The boiling two-phase flow can be in a highly nonequilibrium state; considerable volume of vapor can coexist with subcooled liquid. The amount of vapor that can exist in a subcooled flow at a particular section of the tube is based upon the net result of the rate of evaporation (formation of bubbles) at the wall plus the vapor coming into the section minus the rate of condensation (collapse of bubbles) in the stream.

Suppose there is a flow of subcooled liquid coming into a pipe. Calculations based upon thermodynamic equilibrium dictate that a heat input (namely, Q_{sub} (Btu/hr)) is required to bring this flow to the state of saturated liquid. But, if part of this heat input Q_{sub} is diverted to evaporate some of the liquid at the expense of the sensible heat of the liquid, the increase of the volume-flow rate of the flow will be

$$\frac{\Delta V}{V_T} = \frac{C_l \Delta T_{sub}}{\lambda} \frac{\rho_f}{\rho_v}$$

where V_T is the total volume-flow rate, C_l is the specific heat of the liquid, ΔT_{sub} is the subcooling of the liquid flow coexisting with the vapor flow, λ is the latent heat, and ρ_f and ρ_v are the flow density and the vapor density, respectively. For instance, if the sensible energy represented in 1° of subcooling is used to evaporate water at 1 atmosphere near the heating surface, the percentage increase of volume-flow rate $\Delta V/V_T$ could be 170 percent. This shows how much a flow pattern could be altered if a slight nonequilibrium is allowed. Since the whole boiling two-phase flow is a quasi-steady, nonequilibrium process, it is not unusual to have a significant amount of subcooling in the bulk, and the quality could be much higher than that expected from thermodynamic equilibrium.

Thus the degree of nonequilibrium depends upon the difference between the rate of evaporation at the wall and the rate of condensation in the mainstream; each of these, in turn, depends upon the heat flux, the flow condition, and so forth. All these arguments are aimed at showing that a highly nonequilibrium state can exist for the two-phase flow (the degree being dependent upon the heat flux, among other things). Thus a change in heat flux would alter the quality of the flow. This is probably one reason why the flow patterns of runs B and D are different. In fact, there are many runs in which the total heat input was not enough to bring the flow to equilibrium saturation condition, yet a significant amount of vapor was observed to exist. This shows the importance of recognizing the existance of a nonequilibrium state in two-phase flow.

SUMMARY OF RESULTS

The present study, although preliminary in nature, has shown that the boiling two-phase flow has some unique characteristics that are not encountered in the nonheating two-phase flow. The more important findings are:

1. Whether there are large numbers of active nucleation sites or not determines the flow pattern. When there are large numbers of sites, the flow will go from the bubbly-flow regime, through the coalescence mechanism, to the slug-flow regime. However, when there is a scarcity of sites, there is only a very short bubbly-flow section or none at all. Generally, transition from the bubbly flow to a cavitation slug flow will be rapid.

2. Bubbles leaving the heating surface are ejected into the mainstream at a high velocity. This initial momentum carries some of the bubbles across to the other side of the tube.

3. The trajectories of bubbles can be correlated by analogy to the discharging of jets normally into the mainstream of a steady flow.

4. In the bubbly flow, the growing and departing bubbles disturbed the surrounding flow field so much that the bubbles already in the mainstream were knocked around violently. Apparently the turbulence level of the flow was enhanced by this action. In slug flow, the bubbles forming in the liquid film created ring-shaped ripples that were superimposed upon the capillary ripples usually found on the liquid film during slug flow.

5. The finding by Griffith and Wallis as to the effect of the rate of addition of vapor phase from a channel wall on the flow pattern has been substantiated. In this study, runs with same flow rate and total heat input, but various heat fluxes, were carried out. The flow pattern changed from bubbly flow to slug flow as the average heat flux decreased and the length of heating section

increased. The flow pattern was affected by both the heating length and by the total heat flux. It is also postulated that the extent of thermodynamic nonequilibrium is affected by the ebullition process.

6. The two-phase flow with heat addition appeared to be accelerating; the effect of this accelerating flow on the boundary layer should be assessed.

7. This study shows that the boiling two-phase flow is far more complicated than either pool boiling or the nonheating two-phase flow alone.

Lewis Research Center
National Aeronautics and Space Administration
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APPENDIX - SYMBOLS

C	specific heat
D	diameter
g	local acceleration due to gravity
Q_{sub}	heat required to bring fluid from subcooled condition to saturation
q	heat flux per unit area
T	temperature
ΔT_{sub}	subcooling of liquid flow coexisting with vapor flow, $T_{sat} - T$
u_l	superficial velocity of liquid
u_v	superficial velocity of vapor
u_∞	mainstream velocity
u^*	dimensionless velocity, $u/\sqrt{Dg}$
V	volume-flow rate
ΔV	increase of volume-flow rate
v	radial velocity
x	distance downstream along tube axis from bubble-nucleation site
y	distance perpendicular to wall
y_p	maximum penetration distance of bubble
ρ	density
ρ_f	mean density of fluid in two-phase flow
λ	latent heat of evaporation

Subscripts:

av	average
b	bubble (at departure)
j	jet

local	local
l	liquid
sat	saturation
T	total
t	tube
v	vapor

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TABLE I. - CONDITIONS FOR THE VARIOUS BUBBLES SHOWN IN FIGURE 5

Date	Run number	Symbol in figure 5	Velocity of bubble at departure, v_b, in./sec	Diameter of bubble at departure, D_b, in.	Mainstream velocity, u_∞, in./sec
2-16-62	5-1	□	8.0	0.038	8.3
↓	5-2	○	26.0	.037	↓
	5-3	△	16.2	.033	
	5-4	◹	16.6	.031	
	5-5	▷	9.7	.033	
	3-1	◇	11.1	.025	9.5
	3-2	▽	9.6	.027	↓
	3-3	◿	11.3	.033	
	3-4	◸	14.1	.032	
	3-5	◁	10.5	.031	

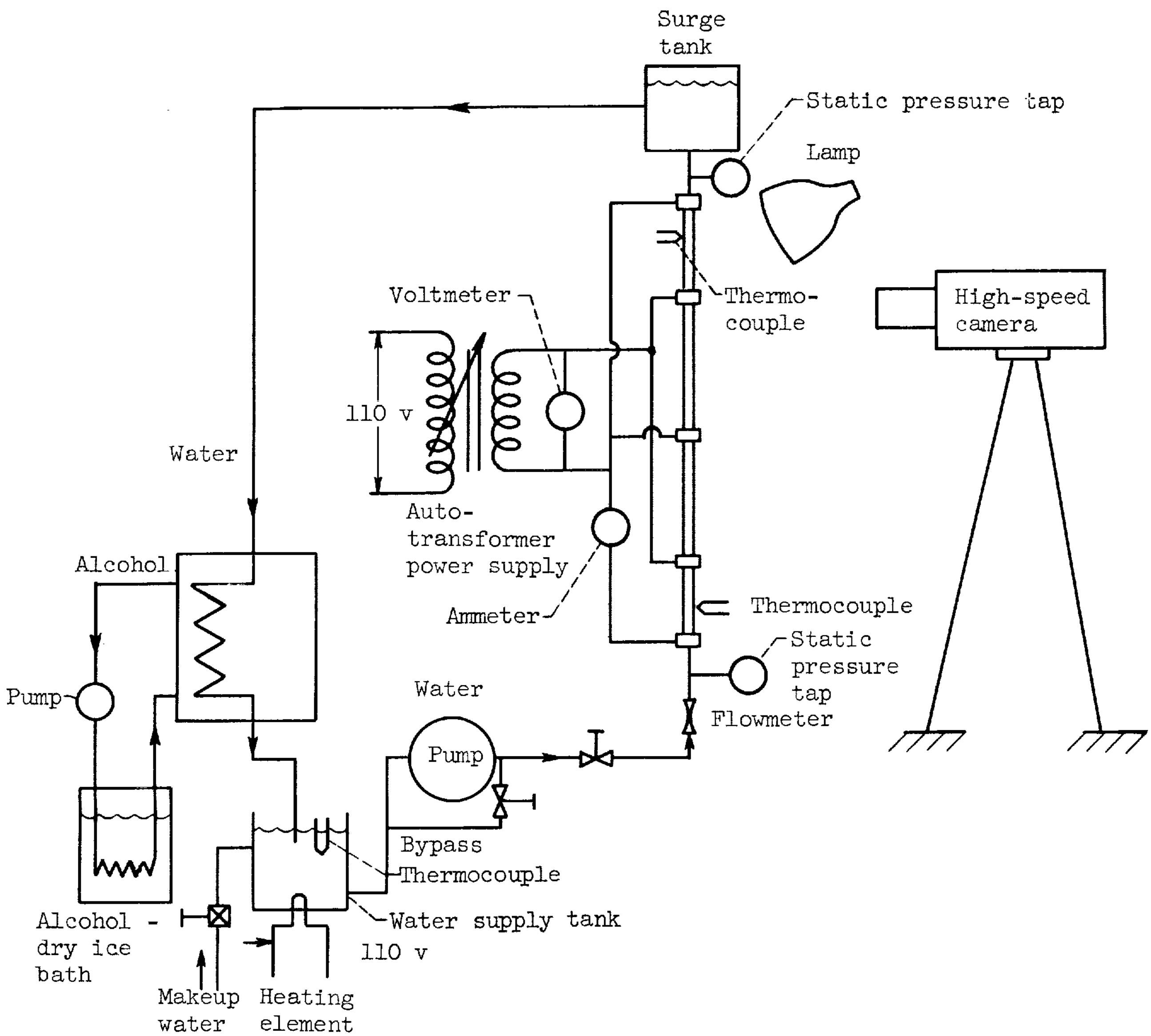
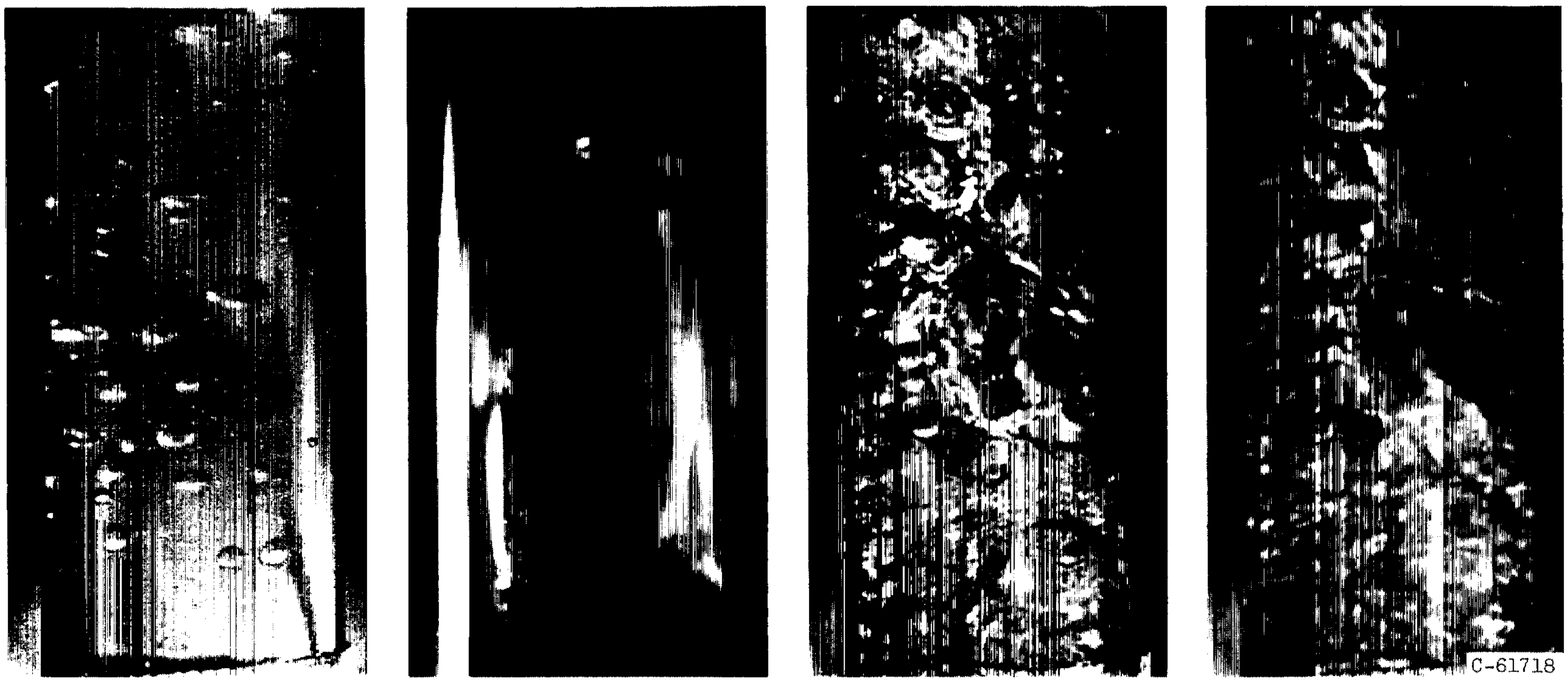


Figure 1. - Test apparatus.



Bubbly flow

Slug flow: No active sites; Active sites

Advanced slug flow developing into annular flow

Figure 2. - Flow regimes observed.

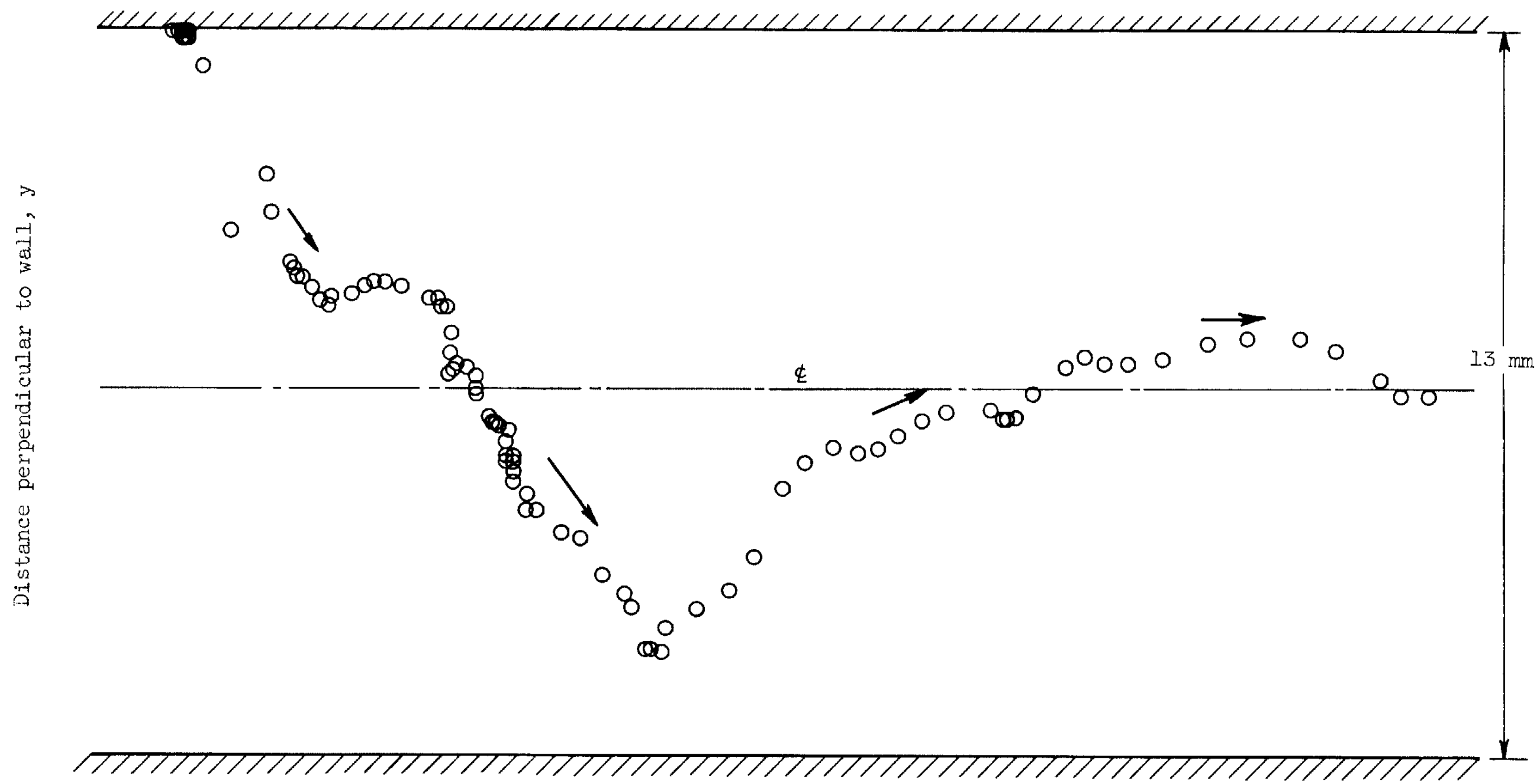


Figure 3. - Bubble trajectory in small-diameter tube.

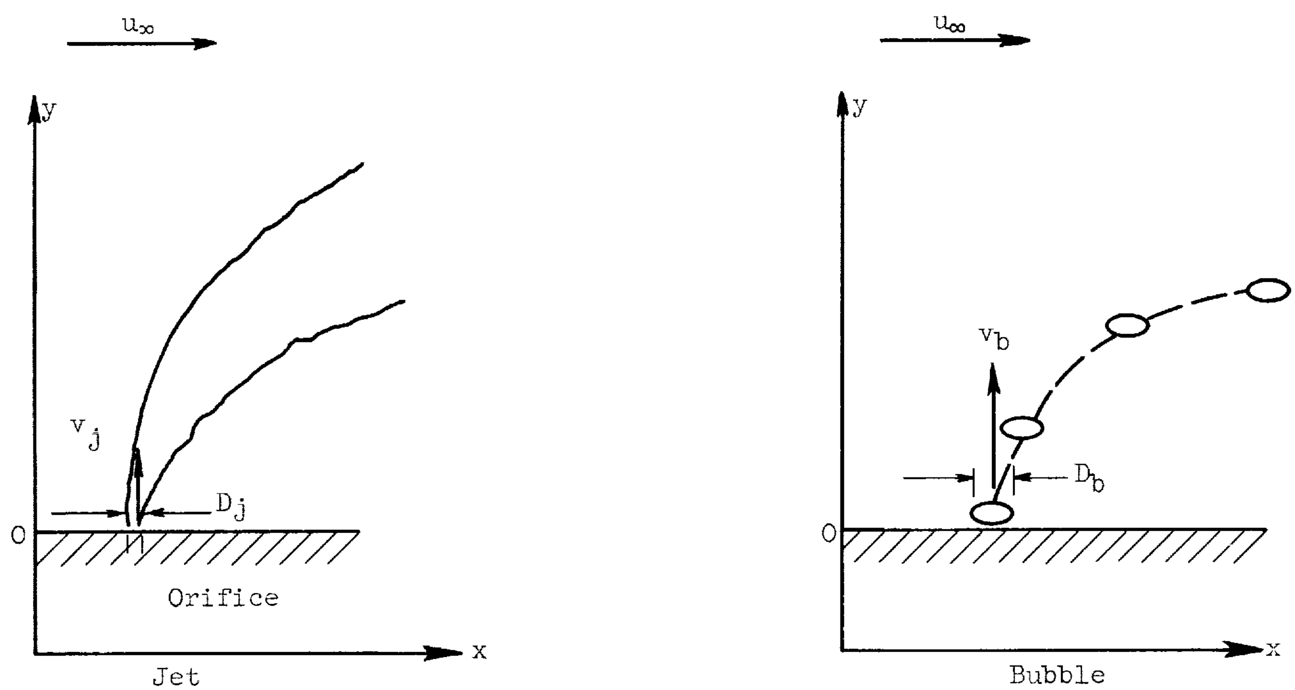


Figure 4. - Analogy between jet penetration and bubble penetration in stream.

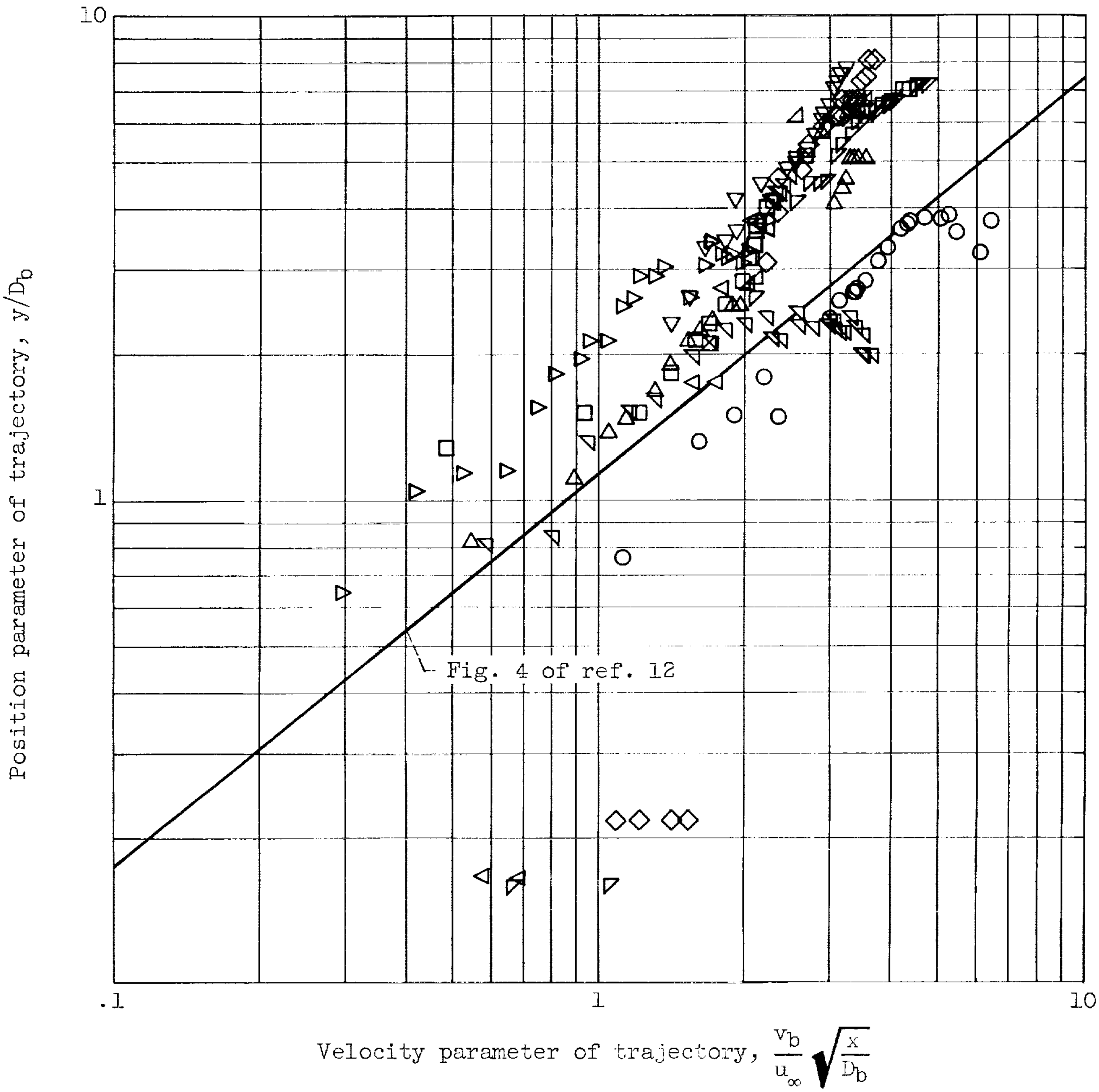


Figure 5. - Comparison of bubble trajectories with jet trajectories. (See table I for definition of data points.)

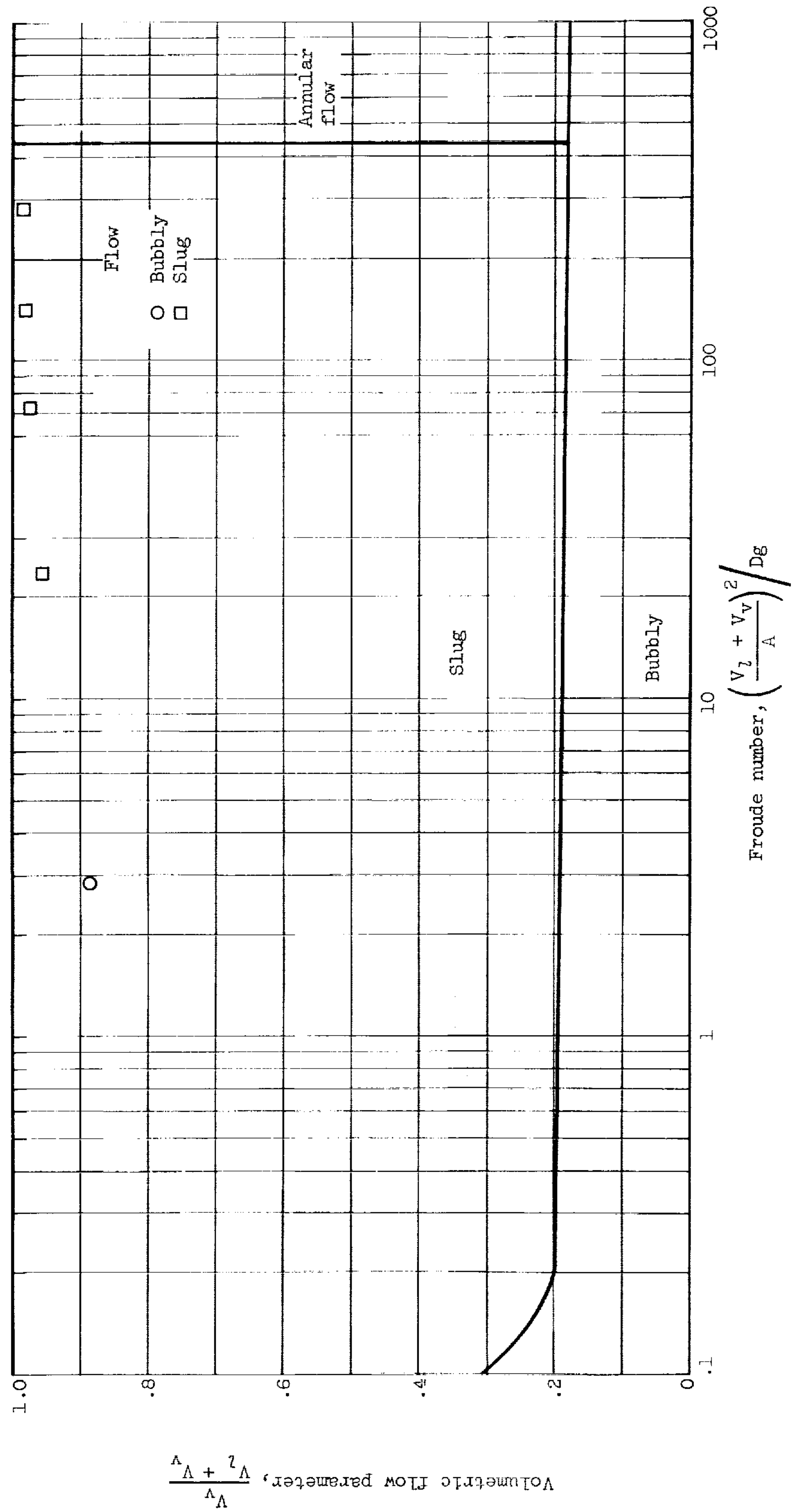


Figure 6. - Map of two-phase flow regimes (ref. 13).

NASA-Langley, 1963 E-1773